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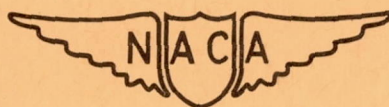
TECHNICAL NOTE

No. 1313

AN INVESTIGATION OF THE EFFECT OF BLADE CURVATURE
ON CENTRIFUGAL-IMPELLER PERFORMANCE

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and Dean M. Dildine

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SUMMARY

Three centrifugal impellers, the same except for angular blade curvature, were investigated to determine the effect of the distribution of blade loading on impeller performance. The blade curvatures are geometrically definable shapes and the differences in blade loading were compared on the basis of particle travel with constant axial velocity along the blade surface. Impeller A has a parabolic blade curvature and a constant blade loading. Impeller B has an elliptical blade curvature with a loading that decreases with increasing impeller depth and that has a higher initial value than that of impeller A. Impeller C has a circular blade curvature with the highest initial blade loading, which also decreased with increasing impeller depth. The blade curvatures of impellers A and B extend the full depth of the impellers; the blade curvature of impeller C extends 0.60 of the impeller depth. The blades of impeller C have a 7.5° forward inclination at the impeller discharge. The three impellers were investigated with a vaneless diffuser in a variable-component compressor test rig with the same conditions and instrumentation.

Impeller B had the highest peak adiabatic efficiency at all equivalent tip speeds except 1400 and 1600 feet per second. Impeller C had the lowest peak adiabatic efficiency of all three impellers at equivalent tip speeds above 1000 feet per second but the largest slip factor of all three impellers at all equivalent tip speeds. The variation of pressure ratio at peak adiabatic efficiency with equivalent tip speed was more nearly the same for the three impellers than the variation of slip factor and peak adiabatic efficiency. Impeller C had the largest maximum specific capacity at all speeds; impeller B had nearly the same maximum specific capacity as impeller C; and impeller A had from 6 to 11 percent lower maximum specific capacity than impeller C. Impeller B had the largest operating range above an adiabatic efficiency of 0.70 at a pressure ratio of 1.80, and impeller A had the largest

This report presents a comparison of the performance of the three impellers over a range of tip speeds and volume flows. The performance characteristics compared include the efficiency, the slip factor, the maximum flow capacity, the range of flow, and the pressure ratio.

IMPELLER DESIGN

Each of the three impellers used in this investigation has 18 blades, the same blade-entrance angles, and the same entrance and discharge diameters. The blade-entrance angle is defined as the complement of the acute angle between blades at the entrance edge and a plane perpendicular to the axis of rotation. The blade-entrance angle for each of these impellers is 60° at the blade tip and the tangent of the angle varies linearly with impeller radius.

An axial-plane section for the three impellers (A, B, and C), with the pertinent dimensions given, is shown in figure 1. Impellers A and B have a radial entrance edge and impeller C has an entrance edge that deviates from the others as shown by the dotted line in figure 1. The indicated difference in the entrance edge of impeller C from that of impellers A and B is the result of the necessity for keeping the entrance angles of all three impellers the same at all radii. The entrance edges of all three impellers were sharpened to a maximum radius of 0.024 inch. Front views of impellers A, B, and C are shown in figure 2.

The blade curvature of impeller A is illustrated in figure 3. Figure 3(a) shows a cylindrical section that has a diameter equal to the outside diameter of the impeller and a length equal to the axial length of the impeller. The blade surface (shaded) consists of radial elements and the intersection of the cylinder with the blade surface defines the blade curvature. The portion of the blade surface used by the impeller is shown in figure 3(b). The bounding axial-plane curvatures were chosen to prevent excessive adverse pressure gradients at equivalent tip speeds below 1200 feet per second.

The passage area, taken perpendicular to a mean-flow-path line formed by rotating each radial blade element into an axial cross-sectional plane of the impeller, was constant.

The blade curvature of impeller A (fig. 2(a)) corresponds to that of a parabola on the developed surface of the cylinder. This

schematic view of this apparatus is shown in figure 4. The collector casing is enclosed in an insulating box, the walls of which are composed of one-quarter inch of hard asbestos, 1 inch of insulating board, and one-half inch of plywood. The impellers were driven with a 1500-horsepower induction motor through a step-up gear. The collector casing was mounted separately from the gear box to reduce heat transfer between it and the gear drive. The arrangement of the equipment is shown in figure 5.

The impellers were tested in conjunction with a 24-inch-diameter vaneless diffuser of constant area. Every component part except the impeller was the same for all tests.

Instrumentation

The equipment was instrumented with thermocouples and pressure taps and tubes located in conformance with reference 3. This instrumentation provides an entrance measuring station 2 pipe diameters upstream of the impeller, and two discharge measuring stations 12 pipe diameters downstream of the collector casing. In addition to the standard instrumentation, pressure and temperature measurements were taken at the diffuser discharge. Three shielded total-pressure rakes composed of five tubes each were equally spaced around the diffuser circumference for the pressure measurements. These rakes were insensitive to angle of yaw up to $\pm 44^\circ$. Two calibrated thermocouples were placed midway between the front and rear diffuser surfaces 180° apart for the temperature measurement. These thermocouples were a high-recovery type, insensitive to angle of yaw up to $\pm 20^\circ$ and similar to the Pratt & Whitney probe described in reference 4. The total pressures and temperatures measured at the diffuser discharge were used for the comparative performance of the impellers in this investigation.

Each impeller is charged with the pressure loss that occurs in the diffuser but not with the loss in dynamic pressure that occurs in the large collector chamber. The temperature measurement at the diffuser discharge was used because it has been found that this procedure allows the performance to be correlated at all inlet-air temperatures, whereas the performance based upon discharge-pipe temperatures does not correlate at all inlet-air temperatures. At ambient inlet-air temperatures, the efficiencies based on the diffuser-discharge temperatures were as much as 0.04 lower than those based on temperatures measured in the discharge pipe.

RESULTS AND DISCUSSION

Impeller Efficiency

The variation of peak adiabatic efficiency with equivalent tip speed for the three impellers is compared in figure 6. At an equivalent tip speed of 900 feet per second, impeller A had a peak adiabatic efficiency of 0.78, and the peak adiabatic efficiency decreased to 0.77 at an equivalent tip speed of 1100 feet per second, to 0.69 at 1400 feet per second and to 0.56 at 1600 feet per second. The rate of change in peak adiabatic efficiency was greatest between equivalent tip speeds of 1400 and 1500 feet per second for impeller A.

At an equivalent tip speed of 900 feet per second, impeller B had a peak adiabatic efficiency of 0.82 and the peak adiabatic efficiency decreased to 0.78 at an equivalent tip speed of 1100 feet per second, to 0.67 at 1400 feet per second, and to 0.53 at 1600 feet per second. The rate of change in peak adiabatic efficiency for impeller B was greatest between equivalent tip speeds from 1300 to 1400 and from 1500 to 1600 feet per second.

At an equivalent tip speed of 800 feet per second, impeller C had a peak adiabatic efficiency of 0.80 and the peak adiabatic efficiency decreased to 0.75 at an equivalent tip speed of 1100 feet per second, to 0.59 at 1400 feet per second, and to 0.54 at 1500 feet per second. The peak adiabatic efficiency of impeller C decreased abruptly between equivalent tip speeds of 1200 and 1300 feet per second.

Impeller B, with the elliptical blade curvature, had the highest peak adiabatic efficiency at all equivalent tip speeds except 1400 and 1600 feet per second. Impeller A, with the parabolic blade curvature, decreased in peak adiabatic efficiency at approximately the same rate as impeller B, and the greatest difference between the peak adiabatic efficiency of the two impellers was 0.04 at an equivalent tip speed of 900 feet per second. At equivalent tip speeds of 1400 and 1600 feet per second, impeller A had the highest peak adiabatic efficiency of the three impellers. Impeller C, with the circular blade curvature, had the lowest peak adiabatic efficiency at equivalent tip speeds higher than 1000 feet per second. Up to an equivalent tip speed of 1200 feet per second, the peak adiabatic efficiency of impeller C was no more than 0.03 lower than that of impeller B, but at equivalent tip speeds of 1300 feet per second and higher, the peak adiabatic efficiency of impeller C was 0.08 to 0.10 below that of impeller B.

The rate of increase in pressure ratio with equivalent tip speed dropped to zero for impeller A between 1400 and 1500 feet per second; it decreased in value for impeller C at 1200 feet per second; and it dropped off increasingly at equivalent tip speeds above 1300 feet per second for impeller B. The variation of pressure ratio at peak adiabatic efficiency with equivalent tip speed was more nearly the same for the three impellers than was the variation of slip factor or peak adiabatic efficiency with equivalent tip speed.

Flow Characteristics

The maximum specific capacity increased with increasing equivalent tip speed at nearly the same rate for all three impellers (fig. 9). Impeller C had the highest maximum specific capacity, which varied from 6850 to 7925 cubic feet per minute per square foot at equivalent tip speeds from 800 to 1500 feet per second, respectively. Impeller B, which had nearly the same maximum specific capacity as impeller C, varied from 6650 to 7904 cubic feet per minute per square foot at equivalent tip speeds from 800 to 1500 feet per second, respectively, and to 7984 cubic feet per minute per square foot at 1600 feet per second. Impeller A had the smallest maximum specific capacity, which varied from 6100 to 7500 cubic feet per minute per square foot at equivalent tip speeds from 800 to 1600 feet per second, respectively. Impeller C with the highest initial blade loading had the highest maximum specific capacity of the three impellers and impeller A with the lowest initial blade loading had the lowest with a maximum specific capacity 6 to 11 percent smaller than that of impeller C.

Operating Range

The performance characteristics of impellers A, B, and C are presented in figure 10, where the pressure ratios at each equivalent tip speed are plotted against specific capacity. Contour lines of constant adiabatic efficiency are also shown on these figures. For a comparison of the performance of these impellers, the useful operating range is defined as the range of specific capacity in which the adiabatic efficiencies are greater than 0.70. These operating ranges are determined from figure 10. The useful operating range at any constant pressure ratio will lie within the 0.70 adiabatic efficiency lines or be limited on the minimum end by the occurrence of surge. The operating ranges at adiabatic efficiencies above

an unstable operating region that was similar in many respects to the unstable operating regions encountered with impellers B and C. Probably stable operating ranges at pressure ratios higher than those limited by surge also exist for the impellers of this investigation. Both the operational instability and the lack of an appreciable operating range occurred only at high equivalent tip speeds; the velocity gradients along the mean flow path of the impeller at these speeds were greater than the maximum value assumed in the design of the passage area of these impellers.

SUMMARY OF RESULTS

An investigation of the effect of angular curvature, or rate of adding angular velocity, of impeller blades has shown for the three blade forms tested:

1. Impeller B, with the elliptical blade curvature, had the highest peak adiabatic efficiency at all of the equivalent tip speeds except 1400 and 1600 feet per second. Impeller A, with the parabolic blade curvature, had the highest peak adiabatic efficiency at equivalent tip speeds of 1400 and 1600 feet per second. Impeller C, with the circular blade curvature, had the lowest peak adiabatic efficiency at equivalent tip speeds higher than 1000 feet per second. For the range of tip speeds investigated, all three impellers decreased in adiabatic efficiency with increasing tip speed.
2. Impeller C had the highest slip factor of all three impellers.
3. The variation of pressure ratio at peak adiabatic efficiency with equivalent tip speed was more nearly the same for the three impellers than the variation of slip factor and peak adiabatic efficiency. Impeller C had the highest pressure ratios at equivalent tip speeds as high as 1200 feet per second, impeller B had the highest at 1300 and 1500 feet per second, and impeller A had the highest at 1400 feet per second.
4. Impeller C had the highest maximum specific capacity at all equivalent tip speeds up to 1500 feet per second. Impeller B had nearly the same maximum specific capacity as impeller C and impeller A had a maximum specific capacity 6 to 11 percent smaller than that of impeller C.

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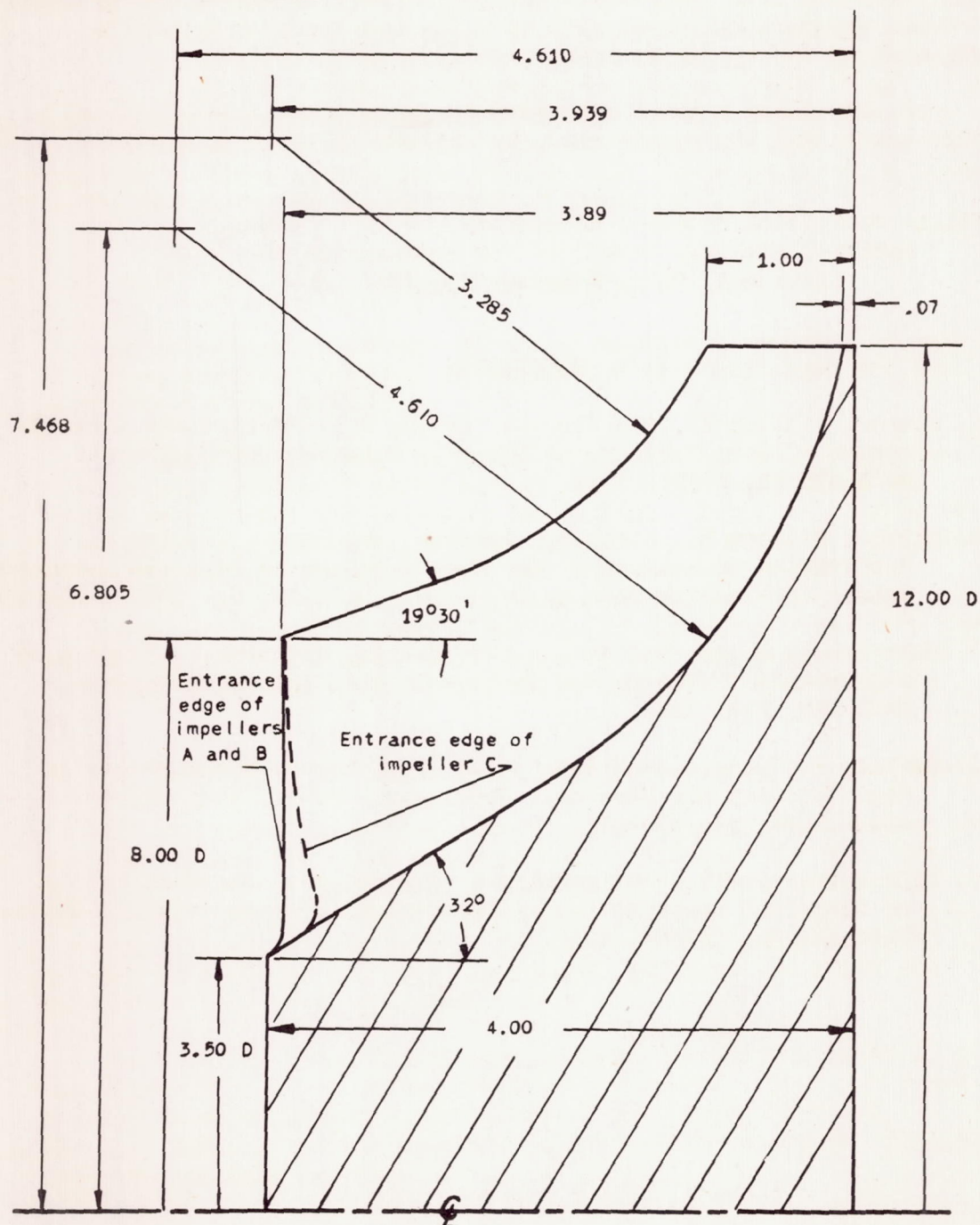
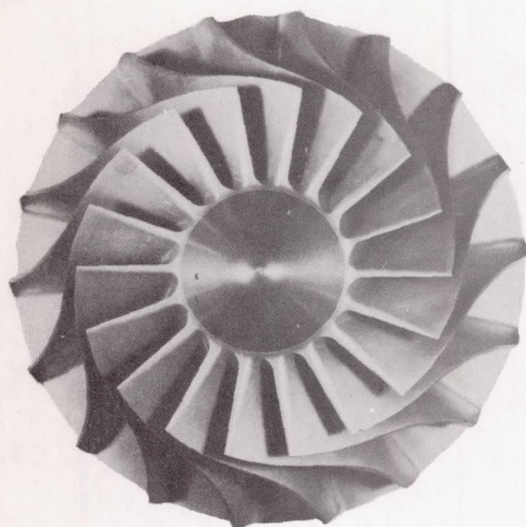
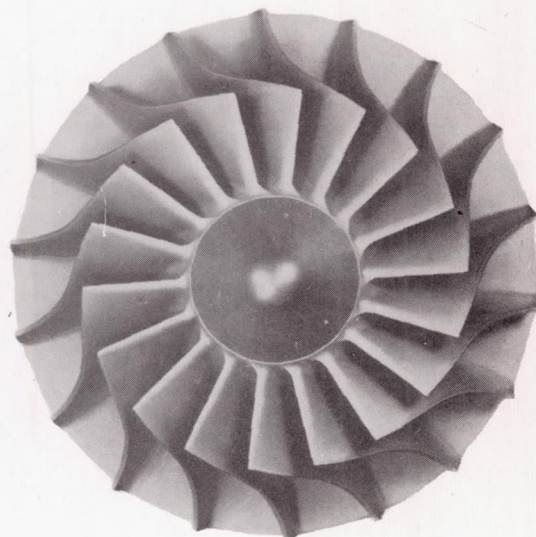


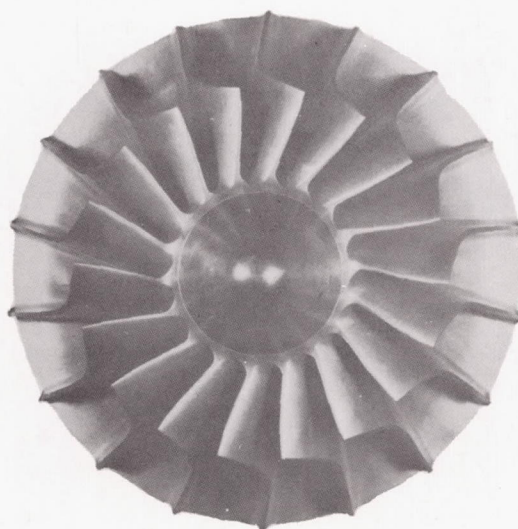
Figure 1. - Axial-plane section of impellers A, B, and C. All dimensions in inches.



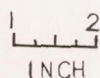
(a) Impeller A, parabolic blading.



(b) Impeller B, elliptical blading.



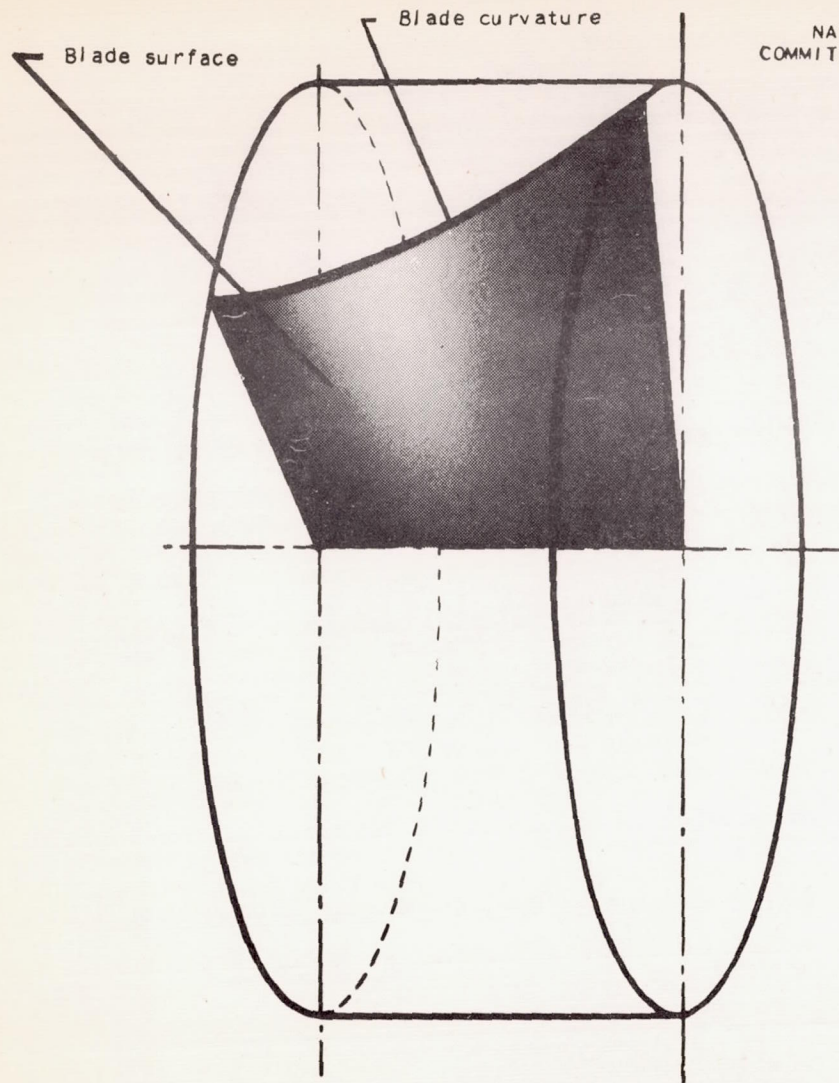
(c) Impeller C, circular blading.



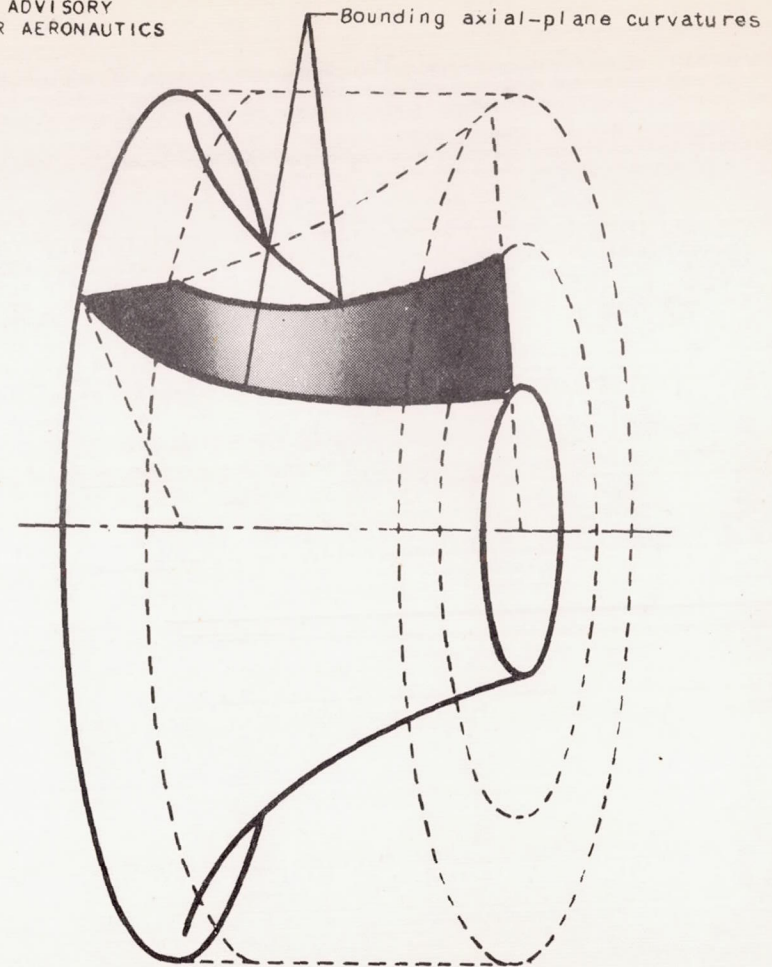
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Figure 2. - Front view of impellers.

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(a) Blade curvature on surface of cylinder.



(b) Portion of blade surface used by impeller.

Figure 3. - Relation of blade curvature to impeller A.

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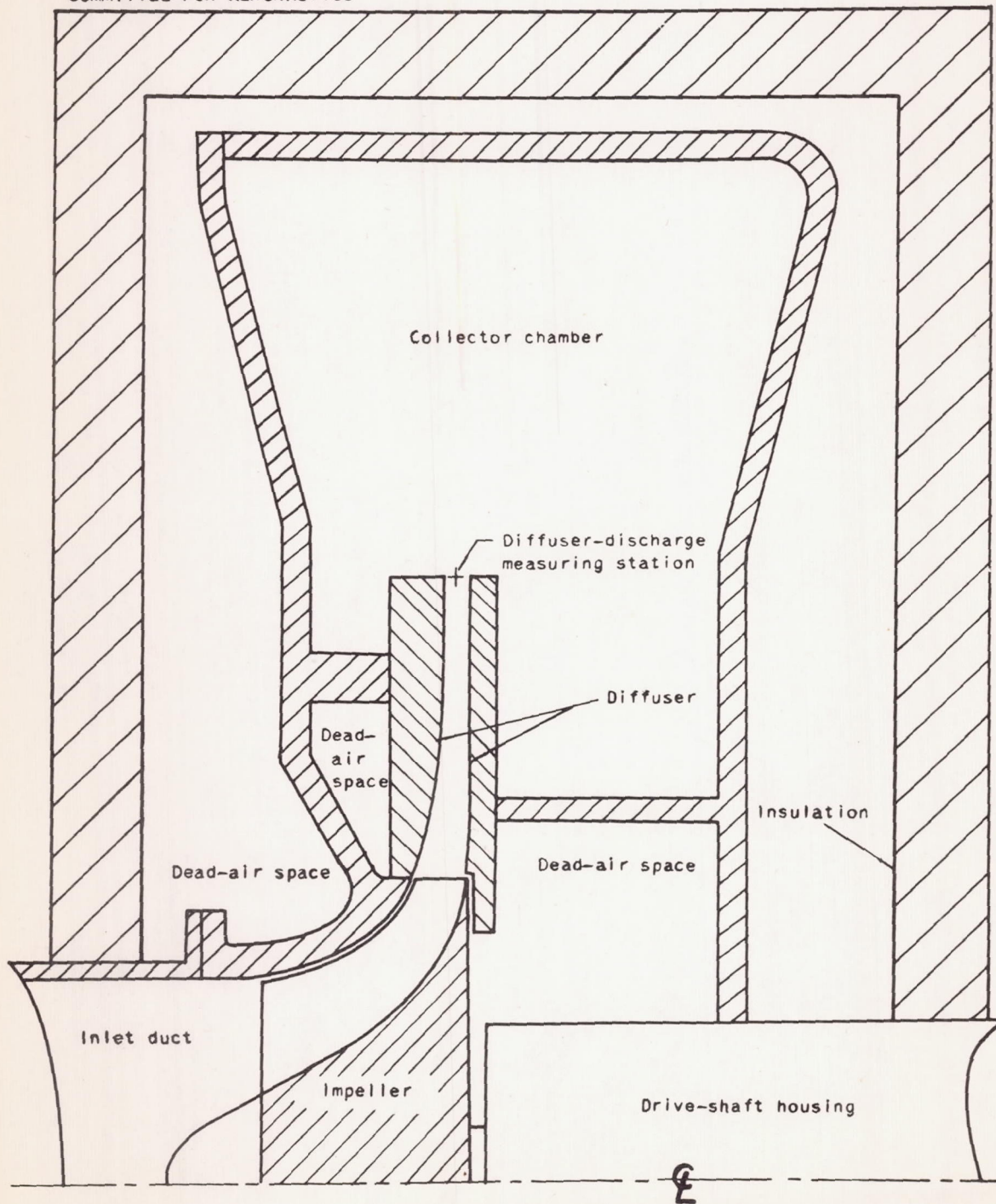


Figure 4. - Schematic view of compressor.

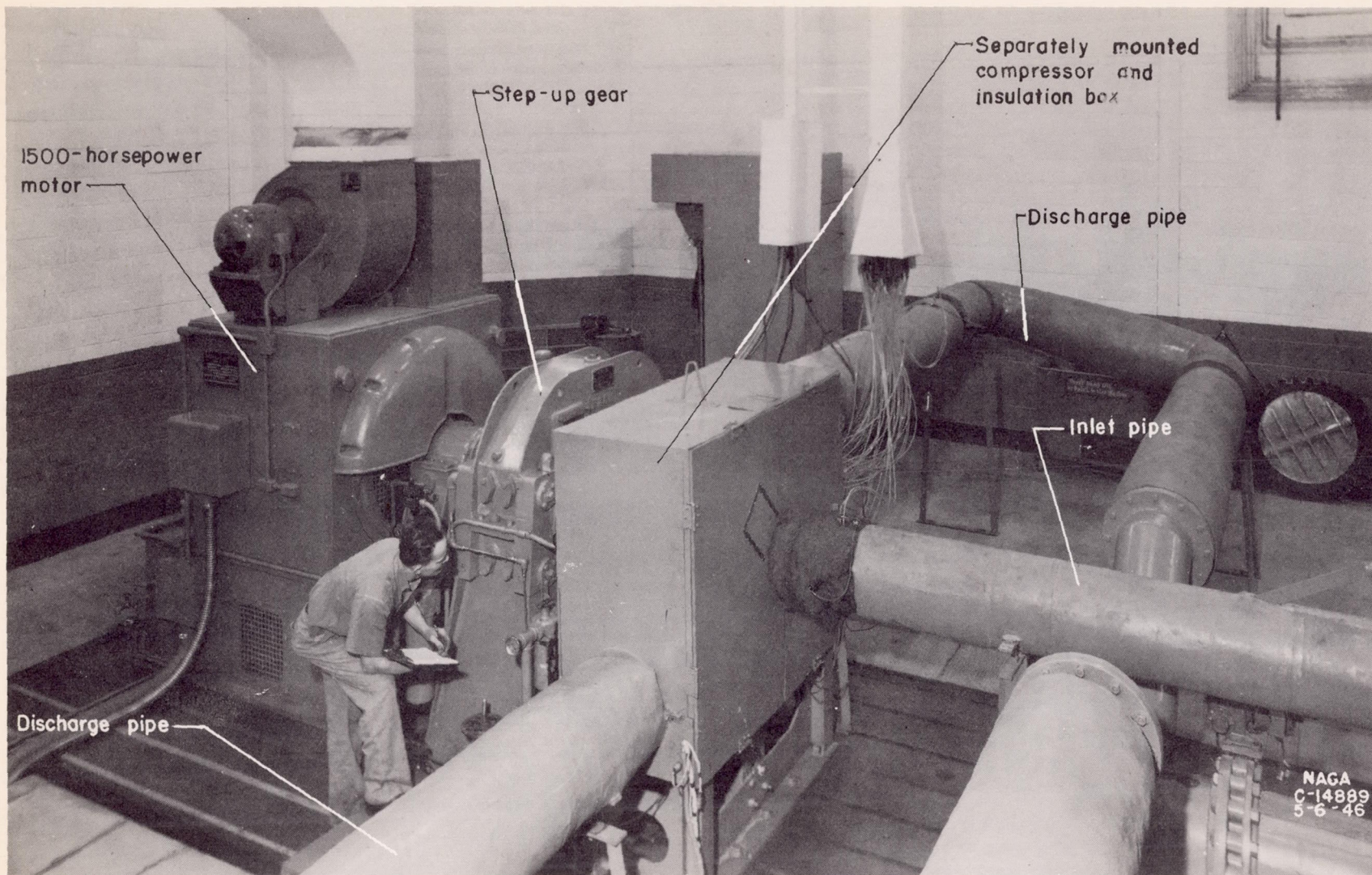


Figure 5. - Setup of equipment.

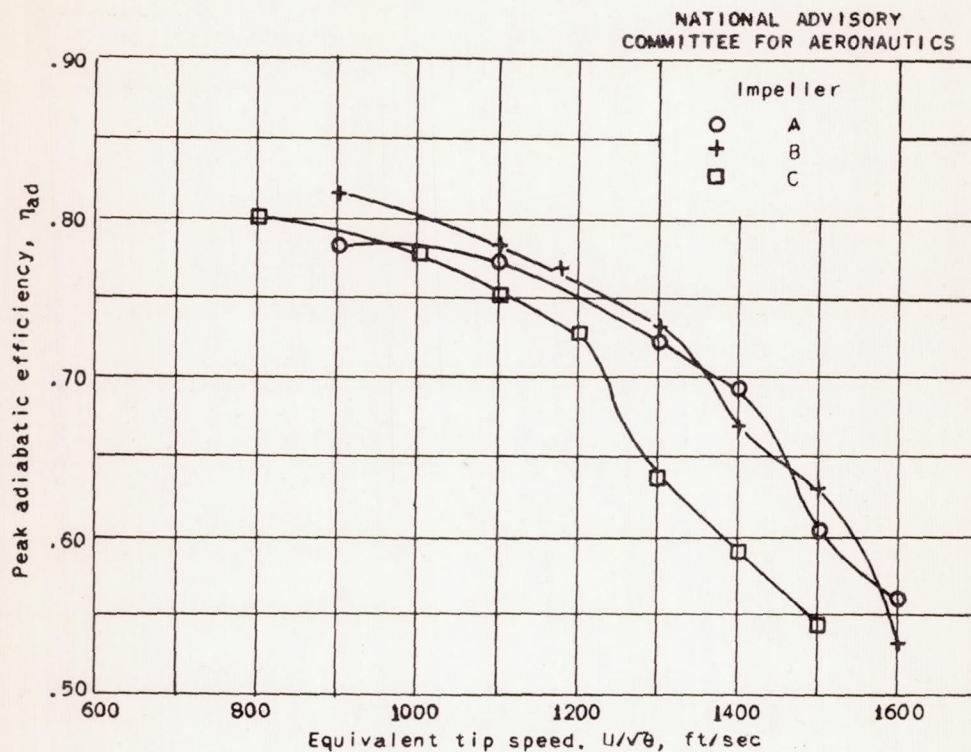


Figure 6. - Comparison of peak adiabatic efficiencies.

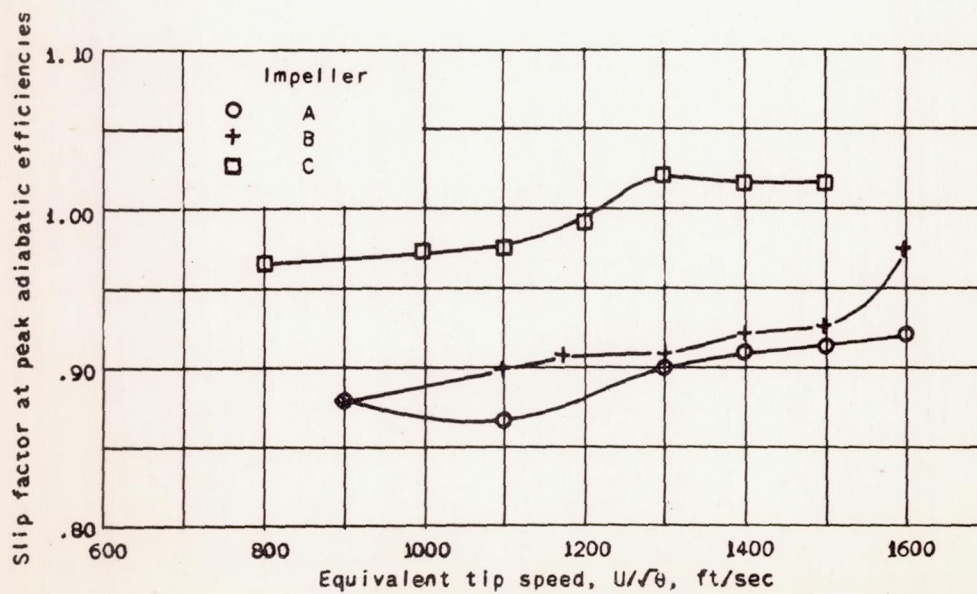


Figure 7. - Comparison of slip factors at peak adiabatic efficiency.

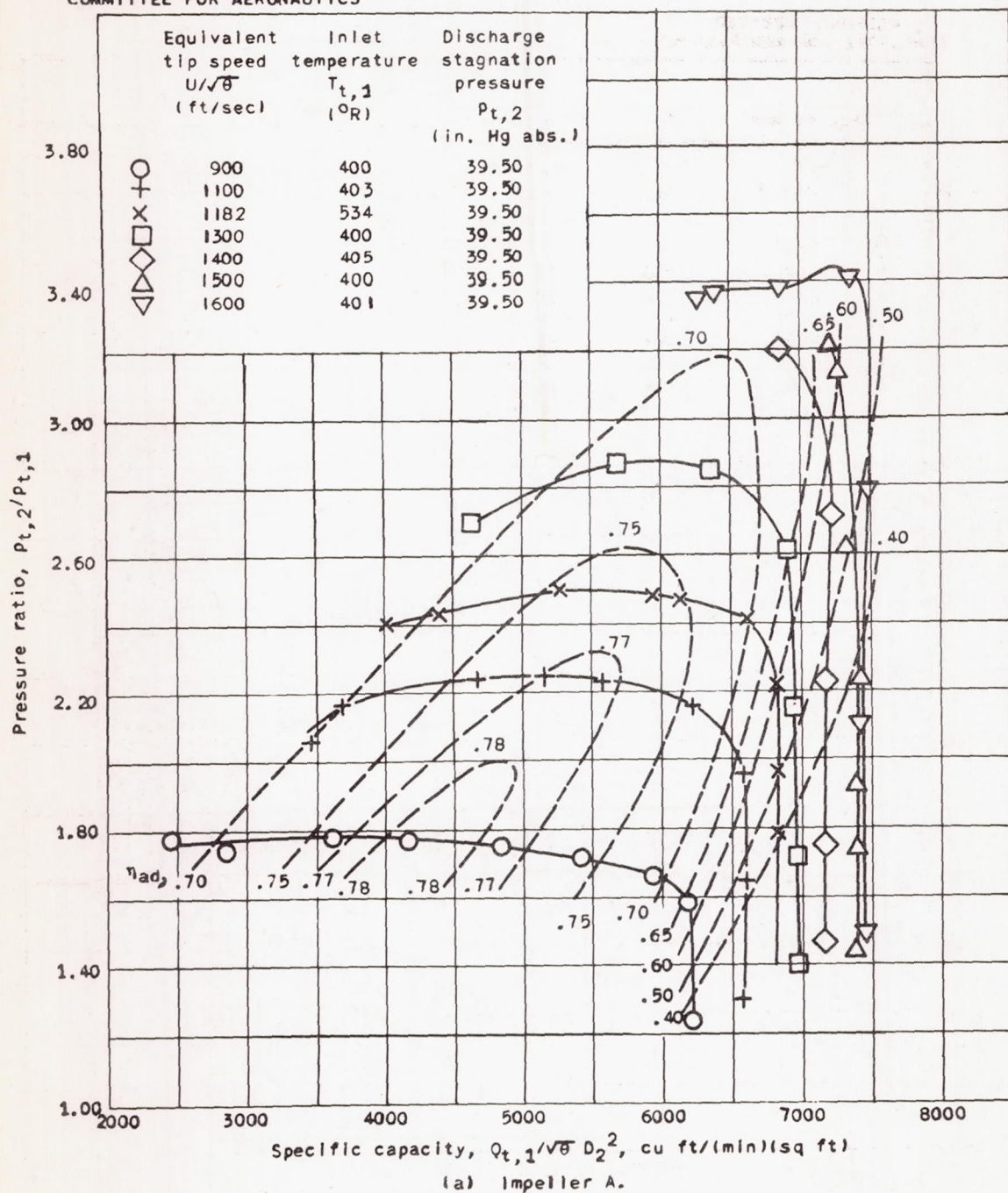
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Figure 10. - Performance characteristics of impellers.

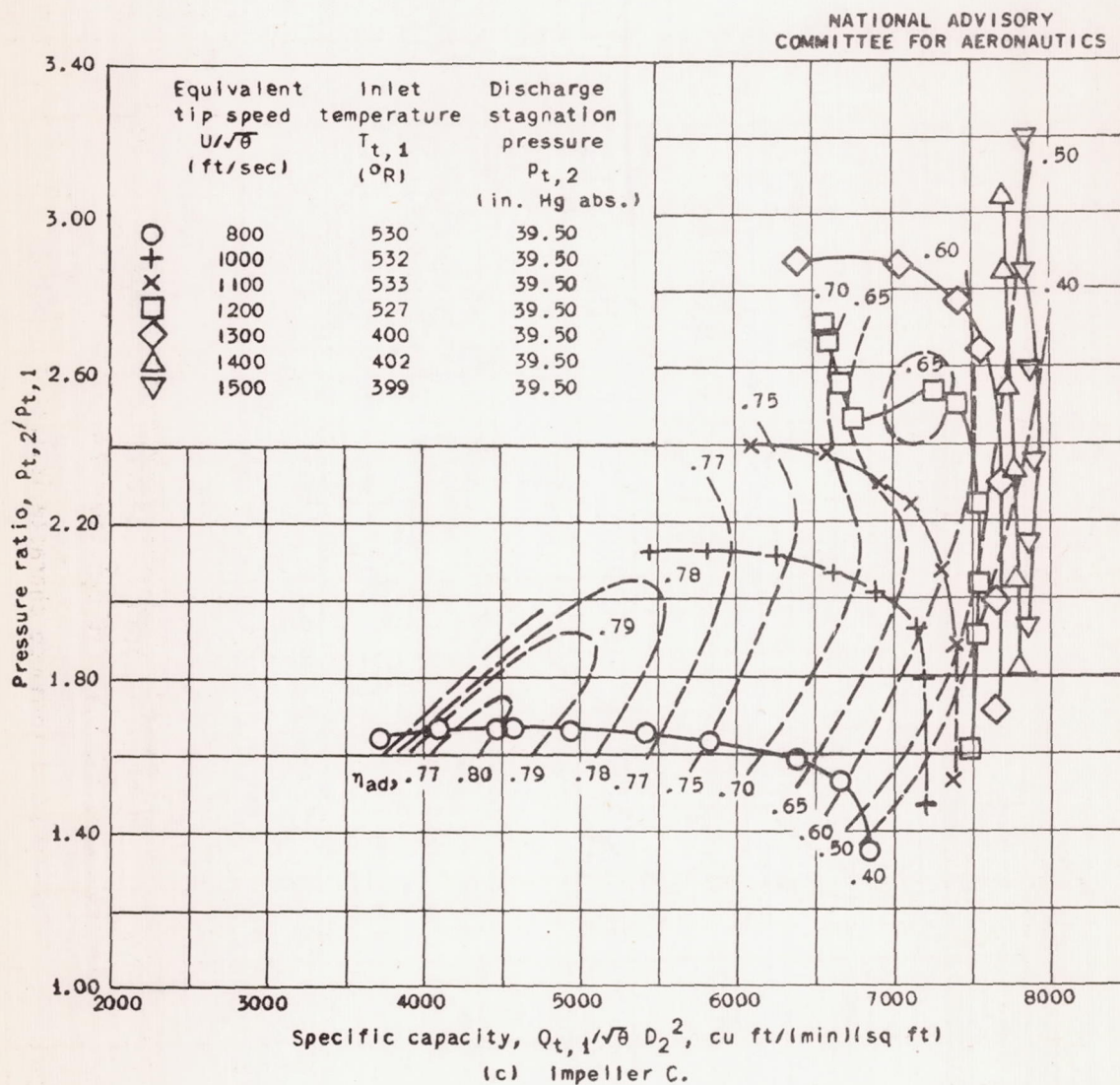


Figure 10. - Concluded. Performance characteristics of Impellers.

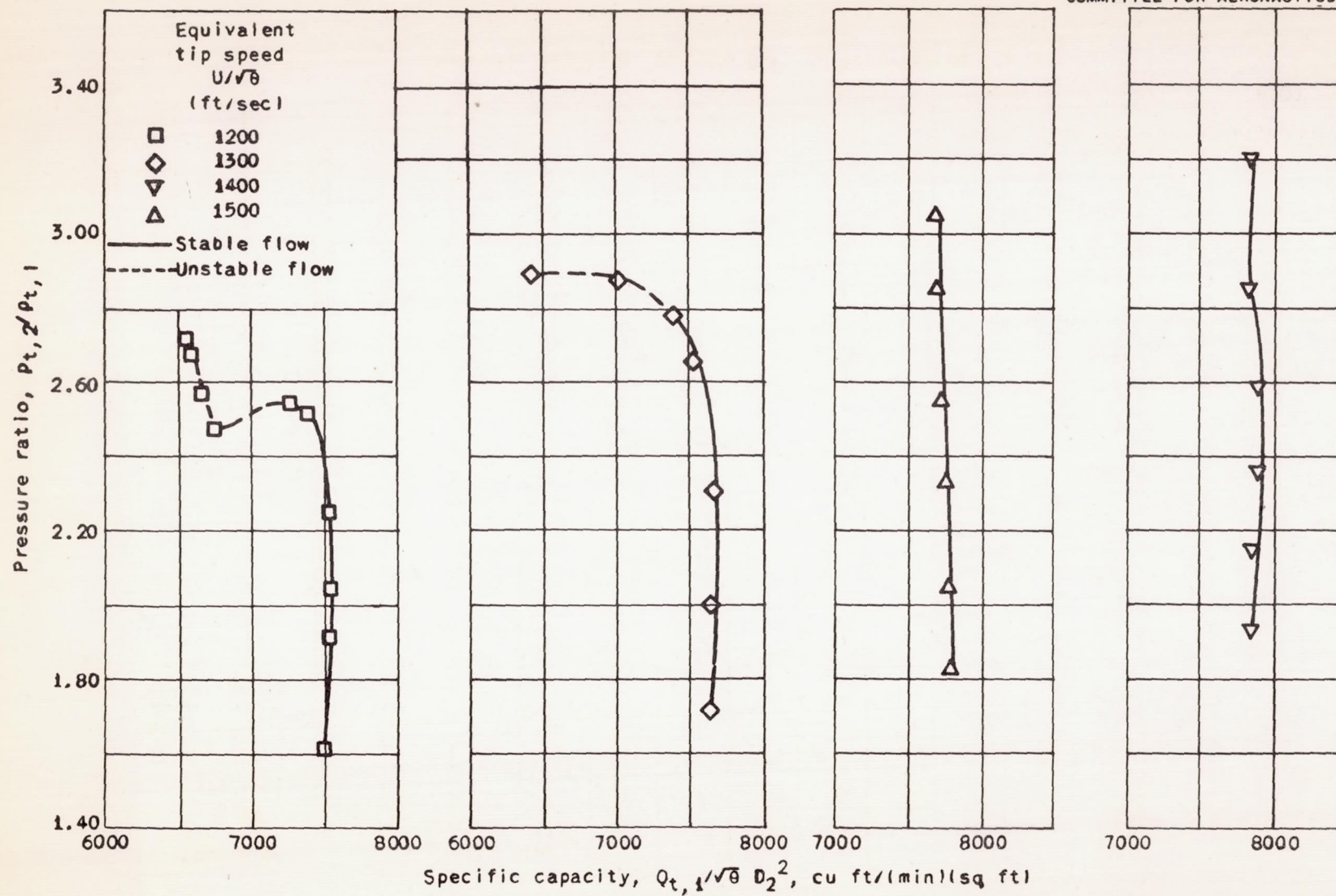


Figure 12. - High-speed performance characteristics of impeller C.